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(54) RECIPROCATING ROTARY ENGINE

(72) Albert, Albert F.,
U.S.A.

(73) Granted to Combustion Research & Technology, Inc.
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RECIPROCATING ROTARY ENGINE

Abstract of Disclosure

Rotary engine having a hollow, stationary block
with manifolds for air inlet and exhaust valving and means
for supplying fuel. The block supports one or more in-
line cylinders which are provided with opposed pistons
5   equipped with rigid and constrained piston rods. The rods
carry bearings that run along a cam track surface interior
to a disc, the outer surface of which is a right circular
cylinder. The surrounding right circular cylinder rotates
as a result of the linear movement of the opposed pistons
10   thereby providing mechanical power. The cam surface is a
continuous track which determines the out-put motion of
the piston movement between top and bottom dead center.
Arcuate areas at top and/or bottom dead center permit con-
stant volume combustion and/or exhaust as desired during
15   a particular cycle, whether that cycle be Otto or Diesel
and whether it be two or four stroke.

Background of the Invention

The invention relates to the art of rotary en-
gines and more particularly to a rotary engine having an
exterior rotary output member driven by opposed pistons
housed within a stationary cylinder or stationary in-line

5   cylinders.

The reciprocal internal combustion engine, as
those skilled in the art are well aware, is the result of
at least 75 years of technological progress. The method
of converting heat energy input to mechanical energy out-

10   put, while regarded today as highly developed, is less
efficient than is possible since conversion of the heat
energy to mechanical energy is done through the piston,
connecting rod and the crankshaft to the rotating output
member. This is true whether the internal combustion be

15   by spark ignition or by compression ignition. The losses
in the system are well recognized and have been extensively
examined over the years. The inefficiency of the connec-
ting rod-crankshaft piston engine is one of the reasons
why a good deal of effort has been turned since World War

20   II to the development of the rotary, turbine and other
types of compact power units.

The constraints imposed by connecting rod, crank-
shaft type engines are numerous. Connecting rod length
and crank radius with their fixed interrelationship to the

25   piston are the primary constraint factors and exert a pro-
found influence on the piston travel function and ulti-
mately on the engine performance. The rapid piston motion,
rise to and fall from top dead center, at speed, present
the greatest problem to the engine designer since ignition

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must occur prior to the engine's achieving full compression
so as to provide time to reasonably complete burning which
continues while the piston moves away from top dead center.
Thus, the connecting rod and crankshaft constraint effec-
tively prevents achievement of constant volume combusiton
of the ideal cycle. A substantial amount of heat is lost
from the combusting fuel or working fluid because of the
relatively slow expansion process which is determined by
the interrelationship of the connecting rod, piston and
crankshaft. Additionally, in conventional reciprocating
piston engines the exhaust valve will be opened before the
piston has reached bottom dead center thereby causing ad-
ditional loss of energy.

In combusting the compressed fuel and thus pro-
viding energy to do work on the piston much energy is lost
through the cylinder head as waste heat. Energy is lost
because of the amount of surface area exposed to direct
flame during combustion process and combustion must be star-
ted before the piston reaches top dead center, causing the
piston to compress an already expanding gas. Energy
dumped into the coolant of a conventional engine, as waste
heat, may be up to as much as 60% of the total energy avail-
able from the burning fuel.

The linear piston movement in the power stroke
is the initial conversion step from heat energy to mechani-
cal energy. The linear motion is in turn converted to the
angular motion of the connecting rod which in turn develops
the circular motion of the crankshaft. Each of these com-
ponents subtracts from the final energy output and is a
design which is completely unbalanced. Balance is achieved
with the addition of compensating weights. Any excess

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weight which must be designed into the engine in the form

of counterweights, flywheels and other compensation fea-

tures and frictional losses associated with the motion of

such parts uses up energy to lessen engine efficiency.

5   As is well known, a four stroke cycle engine

turns its crankshaft 720° from combustion stroke to combus-

tion stroke on a given cylinder. A two stroke cycle en-

gine turns its output shaft 360° from combustion stroke

to combustion stroke. In either the four stroke cycle or

10   the two stroke cycle engine the distance in degrees rota-

tion, from the point of applied energy to the output, is

large since the design of the mechanical system, in the

case of the four stroke cycle, is such that for each posi-

tive power stroke of the piston there are three (3) nega-

15   tive strokes the piston must go through. Further the length

of the connecting rod and the diameter of the crankshaft

are directly related to the stroke of the piston. In net

effect, there has been a great need in the automotive in-

dustry to develop a different engine, initially for higher

20   specific power output per pound of weight and more re-

cently to improve mileage and reduce pollution.

      There are fundamental reasons why the industry

has not leaped into new engine production. Among those

are the fact that most engine cycles lead to larger and

25   more expensive units than conventional power plants and are

of such design that they radically depart from known tech-

nology. This is particularly true of external combustion

engines. Further, of the immediate reasonable alternatives,

such as for example, the Wankle rotary or the gas turbine,

30   each has difficulties. The Wankel which is a competitive

engine for small vehicles does not economically scale up-

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wardly in power, and it is an example of radical departure

from known technology. The turbine which is competitive

for large vehicles does not scale down in power or size

economically. However, the internal combustion engine does

5   not burn fuel completely, is very complex and its mechani-

cal and thermal efficiency leaves much to be desired to

say nothing of its fuel consumption and horsepower loss.

Summary of the Invention

The Almar cycle rotary engine of this invention

10   is provided with a stationary support block of optional

shape which is supported at both ends. Mounted in and on

the block support are at least one but optionally a plura-

lity of aligned, parallel cylinders containing opposed pis-

tons.

15   The fixed support block is hollow so that the

fuel line spark wires and intake and exhaust manifolds

are conveniently directed to each cylinder. The pistons

are provided with rigid piston rods at the outer end of

which is mounted a bearing for engaging with and exerting

20   linear pressure on an inclined inner cam surface of a

surrounding right circular cylindrical rotary output mem-

ber. The rotary output member is rotatably supported on

block journals and is circular around the outside. The in-

side is provided with a continuous cam surface for each cy-

25   linder, which at certain positions of rotation is inclined

to the radial axis of the piston and which is engaged by

the bearings. While the continuous cam surface is not

necessarily an ellipse or a symmetrically continuous curve

it has several consistent and common characteristics. The

30   continuous cam track will provide for two top dead center

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areas of constant volume and, in some cycles, two bottom

dead center areas of constant volume alternated so that

the cam track will be generally suggestive of an ellipse

in which the opposed radii will be symmetrical and which

5   will have a major and a minor axis at right angles to each

other. At top dead center only or at top dead center and

bottom dead center arcuate areas of constant volume radius

from the axis of rotation are provided. The expansion or

power stroke section of the invention, referred to as the

10   Almar engine, will occupy the initial area or general qua-

drant section of the cam which may actually be more or less

than 90° of rotation. The power stroke area may be fol-

lowed by a bottom dead center, constant volume radius area

which will be on or near the long or major axis of the cam

15   curve. In the case of the four-stroke-cycle, the bottom

dead center area of constant volume radius will be followed

by an exhaust section and after that another top dead cen-

ter area radially and symmetrically opposed from the first

top dead center area. The third general quadrant section

20   of the cam track is in an intake section and its profile

is the same as and is symmetrically radially opposed from

the power stroke or expansion section. In the last or

fourth general quadrant section the cam track will provide

for compression back to top dead center and the profile of

25   this fourth section will be the same as and symmetrically

radially opposed to the exhaust section.

        Except in the case where the cam is of a mathe-

matically defined harmonic motion design, at no time does

any point on the profile of the cam surface extend be-

30   yond a straight line which is tangent to the last point

of constant volume radius. No reverse curve such as would

be defined by inwardly extending lobes are provided. Transition areas between arcuate constant volume sections are defined to be smooth for placing the least amount of stress on the bearings and to assure the bearings are in constant

5   contact with the cam surface. There will be a continuous cam surface for each cylinder in a multi-cylinder engine. The axes of each cam will be positioned or rotated with respect to each other so that the mass of the rotary output member remains balanced. It can be readily seen that

10   in the case of the multi-cylinder engine in which the firing order is in rotation, e.g., no two cylinders fire at the same time, the positioning or angular rotational offset of the cams to maintain balance is determined by the number of cylinders. However, in the instance where the cylinders

15   fire at the same time it is necessary to design all cams to be in balance within themselves. The outside of the rotary output member may be provided with gears to be engaged by starter motor and for power takeoff.

         Accordingly, it is among the many features, advan-

20   tages and objectives of the invention to provide a reciprocating rotary engine in which the combustion chamber is formed in conventional, opposed piston configuration which utilizes all the refinements and advances of reciprocating engine technology. The design of the engine allows for

25   the most rapid expansion process possible during the power stroke to reduce heat loss through conduction while converting heat energy to useful work. The engine completes its power stroke in at least one half the time of the conventional connecting rod engine and thus its expansion

30   time will be at least twice as fast. The Almar engine allows for maximum possible expansions to utilize as much

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of the heat energy as possible for useful work, with the
lowest temperature exhaust gases consistent with the chosen
cycle being emitted. The engine can be designed to run
as a complete or extended cycle engine. The engine pro-

5   duces maximum possible pressure prior to expansion without
combustion starting before the piston reaches top dead
center. The increased speed of expansion from top dead
center to bottom dead center results in less heat loss to
the cylinder walls.

10   Because of constant volume combustion, the in-
gested fuel air misture can be ignited after the piston
has reached top dead center. This arcuate area of constant
volume radius can be designed to a specific burn time as
determined by the quality and type of fuel used. The

15   areas of constant volume at bottom dead center in a four
stroke engine allow for maximum expansion of the gases be-
fore the exhaust valves open. This constant volume allows
a dwell time for the piston during which the exhaust valves
are opened thereby relieving pressures after employable

20   heat and pressure are utilized and before the exhaust
stroke is begun. The instant invention causes a more rapid
compression stroke than con-rod engines which results in
less heat exchange time between fuel/air misture and cy-
linder walls thus reducing heat loss. Valve timing be-

25   comes less critical due to the capability of the pistons
to dwell at top and bottom dead center. This is signifi-
cant because it permits more design flexibility in valve
timing and minimizes valve overlap and slower valve rise
and fall. Minimizing valve overlap will give greater

30   volumetric efficiency with less intermixing of exhaust-
intake gases.

The engine is basically free from vibration and is dynamically balanced. The thrust forces are equalized by equal and opposite motion of the opposed pistons. Because of cam design and inherent engine characteristics

5 all acceleration and deceleration transitions are smooth. Vibrations from the piston rod bearing-cam interface are minimized and there is no intermittent loss of contact between cam surface and bearing. The sleeve, friction or frictionless bearing which keeps the piston rod from moving

10 in a direction perpendicular to the linear movement of the piston, is placed as close to the load, i.e., the cam surface, as possible which is at the extreme end of the cylinder wall or support housing.

The engine is basically free from objectionable

15 knock or pre-ignition arising from pressure peak singularities which normally occur as a result of continuing compression after combustion has started and as a result of poor fuel-air mixture.

The period of constant volume at bottom dead

20 center, and the speed of the intake stroke more completely mix the fuel-air mixture which is very helpful in achieving uniformity of the ingested charge prior to combustion and causes the engine to be more tolerant of lean mixtures. Burning of lean mixtures results in better emission manage-

25 ment and less tendency to misfire. Since the combustion chamber is formed between two pistons shaping such chamber to minimize knock is much simpler than in the conventional engine. Heat loss is minimized by the efficiency of the cycle and the speed of expansion as well as the minimized

30 cylinder surface through which heat can be conducted. There will be minimum restrictions on inflow during the intake

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process. The engine will make minimum contributions to
pollution because it is possible to burn very lean mix-
tures in constant volume areas as well as to more completely
burn the fuel.

5   The engine has great design flexibility in that
a variety of cam designs can be chosen to fit the specific
characteristics of a given fuel. A wide variety of cycles
can be chosen with the same basic configuration and the
cam surface configuration can be varied to adapt the pis-
10   ton velocities to a specific end work use. Mass of the
rotating member can be altered within the same basic con-
figuration and the profile of the cam designed to pro-
vide a wide range of piston travel characteristics to op-
timize the time of constant volume and to tailor compres-
15   sion and expansion curves as befits various fuels and/or
cycles. This engine also enables the designer to fire a
multi-cylinder engine in rotation, that is firing cylin-
ders sequentially or to fire some or all of the cylinders
at the same time.

20   The continuous cam track allows the instant
engine to conform more closely to an ideal thermodynamic
curve which of course varies with fuel type, compression
ratio, speed of the engine, type of cycle, etc. The con-
stant volume areas of the cam at top and bottom dead cen-
25   ter may be related to the operating speed of the engine
and/or the burning time of the fuel. The output curve
becomes more closely matched to the curve of combustion
at any engine speed and permits constant volume burning
of the fuel charge for any type of fuel. The rotary out-
30   put member can be designed for high energy storage as a
result of designing weight and speed into the rotary mem-

ber. The engine can be further designed to a particular
intended engine and application, that is high speed low
torque or low speed high torque or constant speed type
engines without compensation features such as torque con-
5   verters. It is an important characteristic that any de-
sired velocity profile and any acceleration/deceleration
curve can be designed into the cam or cams. As a practical
matter and as a consequence of the invention's design
flexibility, it is possible to achieve variances in the
10   acceleration and deceleration profiles or characteristics.
The engine is designed to complete a two stroke cycle with
180° of rotation of the output member, and complete a four
stroke cycle within 360° rotation of the output member.
Because of the opposed piston arrangement and the symmet-
15   rical distribution of the mass in the rotary output mem-
ber there is no need for counterweights or other compen-
sating features for maintaining balance since forces in
the engine are equal. The engine can use as many in-line
cylinders as desired and the cams can be rotated to desired
20   angles, to fire the cylinders simultaneously or to fire
them in series, thus making the engine extremely versatile
in its design capabilities and use applications. The en-
gine is capable or being run with minor modifications from
an external energy supply or external combustion source.

25   Brief Description of Drawings

          Figure 1 is a cross section view of a two cylinder
four piston embodiment showing details of construction and
arrangement of parts;

          Figure 2 is a cross section view along the line
30   2-2 of Figure 1 and further illustrating the details of
the cylinders, pistons, rotor and support manifold;

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Figure 3 is a partial cross section view showing additional details of the structure of a cylinder and piston;

Figure 4 is a perspective view showing how the rotor might look from the outside but without any protective housing around the rotor;

Figure 5 is an additional view showing four cylinders in line;

Figures 6 through 9 illustrate diagrammatically the operation of a two stroke almar cycle, spark ignition engine with opposed pistons as contemplated by this engine;

Figure 10 is a horizontal cross sectional view along the line 10-10 of Figure 11 showing details of a four stroke embodiment of the invention;

Figure 11 is a vertical cross sectional view taken along the line 11-11 of Figure 10 showing additional details of the four stroke Almar engine;

Figure 12 shows representative details of a non-harmonically shaped valve cam;

Figure 13 shows representative details of a harmonically shaped cam;

Figure 14 shows the operational sequence of the four stroke embodiment;

Figure 15 shows general characteristics of a valve cam; and

Figure 16 is a thermodynamic pressure-volume curve showing both ideal and actual curves.

Description of Preferred Embodiment

As those skilled in the art are aware, it is neces-

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sary to understand the constraints imposed on the design

of a connecting rod-crankshaft type reciprocating engine

in order to more clearly appreciate how the engine of this

invention removes such constraints. Essentially the con-

5   straints are the connecting rod lengths from the wrist

pin axis of the piston to the crank axis, and the crank

radius from the crankshaft rotation axis to the crank-

shaft-con rod connection. The relationship between the

linear travel of the piston and the angle of crankshaft

10   rotation is determined by the ratio of connecting rod

length to crank radius. The ratio of connecting rod length

to crank radius has a profound influence on the piston

travel function and ultimately on engine performance. The

rapid rise and fall of the piston to and from top dead cen-

15   ter presents the greatest problem to the engine designer

for it precludes burning at a constant volume at top dead

center and necessitates advance ignition to allow time for

the fuel to burn. For example, an engine operating at

4800 rpm from the time of spark to the end of the fuel

20   burning period takes .002 seconds. This burn time occu-

pies 57° of time rotation. Thus, firing in the conven-

tional con rod crankshaft engine must begin well before top

dead center and obviously results in a direct loss in ef-

ficiency, specific fuel consumption and horsepower.

25   Figure 16 represents a general comparison of

the actual and ideal thermodynamic curves for an Otto cy-

cle and illustrates where the losses occur between ideal

and actual. Such a curve is commonly seen in treatises

which analyze and compare ideal performance with the
actual.* The compression stroke is represented by the line
1-2. An ideal combustion line would be represented by
2-3. The expansion cycle line is shown from 3 to 4 at
5   which time the exhaust ports would be opened and the pres-
sure would drop along the line 4-1. The exhaust stroke
would be represented by 1-0 and the intake stroke by 0-1
and thence back to the compression stroke 1-2. In per-
formance, however, the actual cycle is comparatively repre-
10   sented by the cross hatched portion showing a substantial
loss of efficiency in the engine. The work of the cycle
outside the cross hatched portion is not realized. It is
among other things the requirement to begin burning the
fuel prior to top dead center, the relatively slow expan-
15   sion process during and after combustion, and the opening
of the exhaust valves prior to the piston reaching bottom
dead center that account for the losses resulting in the
actual curve as opposed to the idealized curve.

The advantages of the instant engine accrue from
20   features inherent in its design. The essential mechanical
arrangements of parts and particularly cam design permit
the engine to have the smallest surface to volume ratio
during combustion; the most rapid possible expansion stroke;
the maximum possible pressure at the beginning of the ex-
25   pansion stroke without advanced ignition; and the maximum

*OBERT, "Internal Combustion Engines", p. 497ff, 3rd Ed.,
   International Textbook Company, 1968.
LEWIS, "Gas Power Dynamics", pp. 443-513, Van Nostrand
   Company, New York, 1962.

possible expansion. Finally, the simplicity of the design
lends itself to economical construction and low weight with
the consequent attainment of higher specific power and
overall cost savings because of fewer parts.

5   A two stroke embodiment of the invention is shown
in Figures 1 through 5 and illustrates the construction
and general operating principles of two stroke engines. The
engine, generally designated by the number 10, consists of
a hollow manifold block support member generally designated
10   by the number 12. The member 12 is elongated as shown in
various drawings and is fixed at both ends so that it is
stationary. Internally it is provided in this case with a
partition 14 which as can be seen is slightly off-center
for reasons which will be explained hereinafter. Partition
15   14 divides the inside of member 12 into an exhaust manifold
section 16 and an intake manifold section 18. It will be
quite apparent that intake and exhaust manifolding could be
formed of piping or tubing and directed to the cylinders
without partitioning the interior of the manifold block.
20   Supported by the manifold member 12 are cylinders generally
designated by the number 24. The cylinders have wall 26,
pistons 28, a piston rod 30, rod control bearing mount or
support 32 and threaded cap 34. Rod control bearing 36 is
supported in rod control bearing support 32. It will be
25   noted that piston rod 30 is rigidly attached to the piston
as by threads 38 and is supported by rod control bearings
36. At the outer end of piston rod 30 is secured a roller
bearinal mount 40 in the shape of a yoke supporting roller
bearing 42 on pin 44 which extends into both arms of yoke
30   40. It is to be noted, however, that other kinds of bearings
may be used such as friction, frictionless or slipper types.

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It will be appreciated that the cylinders are provided with
fuel line 46, spark plug 48, exhaust gas ports 50 and air
intake ports 52.

     The rotatable member or rotor generally designated

5   by the number 60 is in effect a series of cam surfaces 62
or any other geometrical combination of curves which will
allow the piston to make a complete stroke within a prede-
termine number of degrees rotation in a single revolution.
The continuous cam surface may be epicycloidal or formed in

10   various symmetrical or asymmetrical configurations resem-
bling an elipse. The exact configuration of the cam will
vary depending, for instance on the degrees of rotation de-
sired to complete a power stroke and the radii lengths in
a particular section of the cam as will be discussed in more

15   detail below. It will be noted that the outside of the
rotor is round and that for balance of weight, the long
axis of one cam surface 62 is in line with the short axis
of the adjoining cam surface 62. In this way, as explained
above, weight distribution in the rotor is balanced and

20   cylinders 24 can be arranged in line in the stationary
manifold support 12. Such angular offsetting of the cams
also contributes to strength in the rotor. The cams are pro-
vided with tracks 64 to engage in this instance, piston rol-
ler bearings 42. The number of cylinders, of course, will

25   depend upon the horsepower desired and such other factors
as the particular application or use of the engine. Rotor
60 has end walls 68 provided with a series of openings such
as is shown in Figure 4. In some designs it might be de-
sirable not to have these openings so that the cam and roller

30   chamber acts as a lubricant reservoir. Bearing or bushing
sections 70 are provided at each end of the engine and are

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journaled on bearings or bushings 72. Gear teeth 74 will
be provided for a power output. Figure 1 shows that a
starter motor 76 with pinion gear 78 will engage ring gear
80 around the periphery of rotor 60.

5   The cam for a two stroke engine will have an area
of constant volume (constant radius) at top dead center the
arcuate area of which may vary according to the type and
burning time of the fuel the particular engine is designed
to use. If the two stroke engine uses ports for manifolding

10   instead of valves there will be no constant volume area at
bottom dead center because of the Kadenacy effect,* that is
the negative pressure created in the cylinder by the opening
of the exhaust port. If the two stroke engine is manifolded
with valves the area of constant volume at bottom dead cen-

15   ter is necessary for the same reasons as pertain to the four
stroke embodiment discussed hereinafter.

          As explained above, the Almar engine is versatile
in that it can be used in two stroke or four stroke mode
in spark or compression ignition. For instance, in Figure

20   5 four cylinders 24 are used, but it will be understood
that one or three cylinders may be used or more than four,
again depending upon the intended application. It will
be appreciated that if an odd number other than one, of cy-
linders are used, such as three, the cam surfaces would be

25   rotated 120° to each other, again to keep the rotor balanced.
In the instance where one cylinder is used or all cams are
aligned and where all cylinders fire at the same time each
cam will be inherently balanced.

*IRVING, "Two-Stroke Power Units", pp. 20 et seq, Hart Pub-
   lishing Co., Inc., New York City, 1968.

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Figures 6 through 9 illustrate an opposed piston
two stroke Almar cycle operation. Cylinder 24 in Figure 6
is shown to be in position for a power stroke by ignition
of spark 48. In Figure 7 the pistons 28 having completed

5   the power stroke open exhaust ports 50 before intake ports
52 are open and thus Figure 7 illustrates the position of
the piston at exhaust. Figure 8 shows pistons 28 as having
finished or completed their outward stroke at bottom dead
center, so that both exhaust ports 50 and intake ports 52

10   are open and thus Figure 8 represents the intake position
of the pistons. In Figure 9 the pistons have closed off
both intake and exhaust ports and thus it represents the
compression stroke.

Figures 10 through 16 illustrate details of a four

15   stroke cycle spark ignition embodiment of the invention.
The engine generally designated by the number 100 includes
stationary manifold block member 102 in which is located a
cylinder generally designated by the number 104. Cylinder
104 includes water or coolant jackets 106 and cylinder walls

20   108. As can be seen cylinder walls 108 extend outwardly to
approximately the circumference or outer radial dimension
of block 102 so that the cylinder has opposed ends 110. Pis-
tons 112 are provided within the cylinders and it will be
seen that each is formed with cavities 114 between which are

25   walls or raised sections 116. Spark plugs 118 are provided
for firing in the chambers defined by opposed recessed areas
114. Each piston is provided with rigid extensions or rods
120 extending outwardly to a piston rod end plate 122. An
intermediate partition 124 located radially relative to the

30   length of the stroke on each side of the cylinder provides
an inner engagement surface 124. A compression spring 126

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is disposed between the intermediate walls 124 and piston

rod end plate 122 to resiliently bias and hold bearings

128 in engagement with the cam surface. Bearings 128

will rotate on shafts 130 and the piston rod or extension

5   120 will be guided by thrust control bearings 132 located

at the outer ends of the cylinder. It is to be noted that

while roller bearings 128 have been shown friction or slide

type bearings may also be used. Bearings 128 will bear on

the cam track generally designated by the number 140.

10   Figure 10 shows additional details of manifolding

and manifolding cams. The rotary output member surrounding

and rotatably mounted on block 102 is generally shown by the

number 134. It includes drive cam member 136 having an in-

ner drive cam surface generally identified by the number

15   140. Rotary output member 134 has cam spacers 142 so that

on one side of drive cam 136 is an intake exhaust valve cam

146. Figure 15 shows that valve cams 144 and 146 are approxi-

mately identical in appearance except, of course, that the

raised portions for compressing and opening the valves will

20   be located differently according to the timing required.

The cam surfaces 148 and 150 respectively of the two valve

cams engage valve stem pushers 152 and 154 which could also

be roller, friction or other types of bearings. If for in-

stance, cam member 146 is the intake valve cam it will en-

25   gage pusher 154 which in turn will depress valve stem 156

against spring pressure 158. Thus valve 160 would be dis-

engaged from its seat to enable combustion gases to enter

the cylinder through manifold channel 162. In like man-

ner on the exhaust side pusher 152 engages cam 144 to de-

30   press valve stem 164 to unseat valve 166 to enable exhaust

gases to be purged through the exhaust manifold channel 168.

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Rotary member side plates 170 are provided together with appropriate bearing seals 172 for rotatably mounting the rotary output member 134 on block 102.

Figures 14 and 15 are included to show that the
5   Almar cam curve configuration is an important feature of
this invention when considered with other design features.
The cam which is driven into rotation by action of the expanding gases working on the pistons during the power stroke
of the cycle also controls piston motion during exhaust, in-
10 take and compression strokes. The cam curvature is designed
to optimize particular features of engine performance for
which certain constraints in cam design are appropriate. A
preferred cam surface is that which permits constant volume
combustion in spark or compression ignited internal combus-
15 tion engines over a wide range of engine speeds, permits the
most rapid expansion of burned gases after energy into
mechanical work, and then decelerates the piston to zero
velocity as rapidly and as efficiently and with as little
wear as possible. Any cam area which for instance would
20 cause bearing 128 to lose contact with the cam surface or
which would result in intermittent bearing contact would be
obviously unacceptable. Also, any cam design which would
cause a large side thrust against the piston rod control
bearing, i.e., the beginning of the exhaust and compression
25 stroke would be unacceptable. Additionally, any portion of
the cam which would give rise to unnecessary problems of
stress and strain, lubrication or which were unnecessarily
complex would also be unacceptable. Preferred surfaces de-
pend for the most part upon the type of engine, fuel, materials,
30 size, specific power, speed, design for load, and other vari-
ables.

The design constraints placed on the cam configuration are set forth in the following general propositions. The cam of Figure 12 illustrates a geometrical limitation on the expansion or power stroke portion of the cam which

5   enables the most rapid possible expansion of the combustion chamber or fastest possible expansion displacement of the pistons in a cylinder. The maximum rate at which the piston moves away from top dead center during combustion is in the acceleration area B of the expansion stroke and is es-

10   tablished by a tangent line T. The maximum value of R_i (index radius) in area B at any point during acceleration of the piston cannot be greater than any point on line T in area B which is tangent to the last radius value R_{KT} (constant radius top dead center) of the constant volume

15   area A. The length of the straight line T of section B extends for as long as acceleration is desired and of course as its transition into area C is smooth. C must be a smooth curve which decelerates the piston until its linear velocity is zero at the beginning of the bottom dead center area D.

20   Again, in following the tangent line T in Section B which is tangent to circular arc A at the last constant volume radius value R_{KT} thereof, the piston accelerates as rapidly as possible while maintaining a positive contact with the cam surface while the pressure between the pistons is smoothly

25   decreasing.

The acceleration area B is shown to subtend 35° but could be larger or smaller and it is not necessary that deceleration area C represent the same number of degrees of rotation as area B. In order to establish cam deceleration

30   Section C it is necessary that it be plotted with the same number of radii as the acceleration curve. Thus, for ex-

ample, seventy-one values of R_i during acceleration are es-

tablished at half degree intervals for the 35° of rotation

of section B. There will be the same number of values for

R_i in the deceleration Section C if it is a larger angle

5   but the index angle for extablishing values of R_i will be

greater than one half degree. If the deceleration Section

C is smaller than acceleration Section B in terms of num-

ber of degrees of rotation the values of R_i for Section C

will be at closer angular intervals than one half degree.

10   In any event and despite the angular intervals the same

values of the changes in R_i will be used in reverse order

to the point at which deceleration Section C joins or reaches

the only or first value of bottom dead center or the constant

volume R_{kb}. Thus, for example, if the last five values of

15   changes in R_i or ΔR_i in the acceleration curve area B are:

Index Number	Delta (Δ) R_i
85	.048746
86	.048880
87	.048941
88	.049048
89	.049082

20   then the first five values of ΔR_i in the deceleration curve

area C are:

90	.049082
91	.049048
92	.048941
93	.048880
94	.048746

25   Accordingly, while the piston during deceleration in cam sec-

tion C is still moving out toward bottom dead center and R_i

is still increasing towards its maximum value at bottom dead

center the differences in R_i or ΔR_i are the reverse of the

radius changes in the acceleration curve B.

30   The cam of Figure 13 represents a mathematical con-

straint on or expression of the acceleration - deceleration

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profile of Sections B and C. Essentially, it is a harmonic curve extending between the last constant volume radius of top dead center and the first or only radius of bottom dead center. In short, the harmonic curve extends through the deceleration and acceleration sections of the expansion part of the cam and is generally defined by the following expression:

$$R_\theta = S + [1-Cos(u)]$$
$$\text{where } u = \frac{\pi}{L} (\theta - k)$$

k = beginning radius point for the harmonic curve (last constant radius at top dead center) in degrees

m = ending radius point for the harmonic curve (first or only radius at bottom dead center) in degrees

$L = 180 - (k + m)$

$S = 1/2$ distance of travel of one piston.

It is to be repeated that the expansion section of the cam may by geometrical design or mathematical expression be greater or less than the first 90° or rotation. Again, any given radius on the cam has an opposite and equal radius so that defining 180° of the cam also defines the other 180° thus giving the cam equal and radially opposed symmetry. The cam does not require that there be an area of constant volume (radius) at bottom dead center since the need for constant volume at bottom dead center will be a design variable.

The exhaust portion of the cam is designed to return the piston to top dead center with a minimum of thrust loads against the piston rod control bearing. This is done by the cam surface being designed so that the linear move-

ment of the piston on the return movement is relatively

slow at the beginning of the exhaust stroke. When the pis-

ton has reached a point at which the distance from the face

of the piston to the control bearing is larger than the

5   distance from the control bearing to the point of contact

at the cam surface, then the cam profile from such point

to top dead center is designed so as to make the linear

movement of the piston very rapid. This type of cam de-

sign will allow for minimum wear on the piston, the piston

10   rods, the cylinder wall and the bearings. The same cam pro-

file for expansion is used during that quadrant which con-

trols the intake stroke. A very rapid movement of the pis-

ton is accomplished during the initial part of the intake

stroke by which great turbulence of the fuel air mixture

15   is created. This gives the desired complete mixing of the

fuel-air ingested charge and again minimizing the wear on

the parts. The compression stroke area of the cam is the

opposite and radial equal of the exhaust section. It will

be appreciated that any two opposing sections or quadrants

20   of the cam will be the same so that the limitation of

equal and opposite radii is not violated.

Figures 14A through 14H depict timed views of a

one cylinder cam in a four stroke Almar cycle embodiment of

the engine. The intake manifold is shown on the left side

25   of the cylinder and exhaust manifold on the right. View A

shows the engine at the beginning of the compression stroke.

At this time the intake valve has begun to close and will

be closed in a very few more degrees of rotation. View B

shows the engine partially through its compression stroke

30   with both valves closed. By the time the engine reaches top

dead center the compression air fuel mixture has been ig-

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nited, is burning, and is creating high pressure in the
clearance space between the pistons. Note that by View C
the cam has traveled only 90° of action rotation. For a
normal engine using a crankshaft, the piston travels from
5   bottom dead center to top dead center through a rotation
of 180°.

On the power stroke View D, the pistons and rods
are seen to drive the rotating cam in a direction to allow
expansion of the hot gases. The cam is driven from opposite
10   sides in a balanced configuration so that there is no net
thrust on the cam in any direction and only radial, equal
and opposite force to make the rotor rotate. Note that
since the pistons are symmetrically oriented and are driven
in unison there is no unbalance of piston forces which
15   would operate to cause lateral vibration. View E shows the
pistons fully extended in the position of bottom dead cen-
ter and the exhaust valve and now open to expel the spent
exhaust gases. Because of constant volume areas the piston
is exerting pressure on the cam throughout its total travel
20   and the exhaust valves do not have to open before the pis-
ton reaches bottom dead center. The cam's angular momen-
tum in the case of a one cylinder engine, now carries the
rotor through this bottom dead center position and begins
to force the gases out of the cylinder during the exhaust
25   stroke. With two or more cylinders or multi-cylinders where
all cylinders are simultaneously fired angular momentum
carries the rotor through to the next firing stroke. The
angular momentum carries the cam through the exhaust stroke
(View F) continuing to expel the gases until top dead center
30   as shown in View G. It will be seen in the Views A through
G that there is always pressure in the cylinders forcing

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the rod bearings to maintain contact with the rotating cam.
During the intake portion of the cycle it is necessary to
force the piston to extend and to this end the spring 126
as seen in other views is mounted between the cylinder and
5 bearing. The springs are always under compression so that
they act to maintain bearing contact with the cam.

WHAT IS CLAIMED IS:

1. A reciprocating, rotary, combustion engine, comprising:

a) a hollow, non-rotatable support block member of predetermined length and adapted to be supported in an operative position,

b) at least one cylinder member supported in said block member such that the ends of said cylinder open to the outside of said block member and whereby within said block member there is space externally of the cylinder, said cylinder including opposed pistons with piston rods extending out of the opposed ends of said cylinder and said piston rods being provided at their outer ends with follower bearing means for engaging a driven cam surface,

c) a rotary output member mounted for rotation around said support block member and including a generally round exterior surface and a continuous interior driven cam surface for said cylinder, pistons and piston rods spaced from said support block, said driven cam surface being engaged by said follower bearing means to rotate said output member,

d) said driven cam being shaped such that it includes predetermined arcuate areas of constant radius from the axis of the rotary member at least for radially opposed top dead center positions for said pistons in order to allow for predetermined periods of time and degrees of rotation at which the chamber between the pistons is at constant volume, said cam also providing at least radially opposed bottom dead center positions, and said cam further

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being so shaped that every radius therein has an equal and opposite counterpart radius,

     e) said cam also being shaped after the top dead center constant volume areas to provide radially opposed rapid expansion areas of a predetermined number of degrees of rotation of the rotary output member in which a first portion of each of said rapid expansion areas moves the piston away from top dead center at an accelerating velocity and a second portion moves the pistons away at a decelerating velocity toward bottom dead center, and

     f) said cylinder being provided with means for admitting fuel thereto, means for timed ignition of the same and further provided with air intake and exhaust gas manifolds and means for selectively opening and closing said cylinder to said intake and exhaust manifolds.

     2. The engine according to Claim 1 and in which the same is shaped after bottom dead center to provide opposed return areas of a predetermined number of degrees of rotation of the rotary output member for returning the pistons to the next top dead center area.

     3. The engine according to Claim 2 and in which said cam is shaped after the top dead center constant volume areas to provide an expansion section having acceleration and deceleration areas and wherein the maximum value of the radius in the acceleration area cannot be greater than any point on a line which is tangent to the last constant radius value of the top dead center area.

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4. The engine according to Claim 3 and in which the deceleration area of the expansion section in limitation (a) is defined as a curve which continues to move the piston outwardly at a decelerating rate and which rate decreases in reverse order of changes in the radii by which the acceleration area increased.

5. The engine according to Claim 4 and in which the expansion section may be of a greater or lesser amount than 90° of rotation in order to extend the pistons to bottom dead center position.

6. The engine according to Claim 5 and in which the exhaust section is shaped after bottom dead center position to return the pistons to top dead center position at first slowly and then increases the speed so as to minimize thrust loads on the piston rods and bearings.

7. The engine according to Claim 6 and in which said driven cam surface is provided with predetermined arcuate areas of constant radius from the axis of the rotary member for radially opposed bottom dead center positions for the pistons in order to allow for predetermined periods of time and degrees of rotation at which the chamber between the pistons is at constant volume at bottom dead center.

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8.   The rotary engine according to Claim 3 and
in which said intake and exhaust manifolds are opened to
said cylinder by ports and which ports are opened and
closed according to the position of the pistons.

9.   The rotary engine according to Claim 3, and in
which said intake and exhaust manifolds are provided
with openings which are provided with and opened and closed
by valve members actuated in accordance with the require-
ments of the engine.

10.   The rotary engine according to Claim 9 and
in which said valve members are provided with stem portions
which engage valve cam means against which resilient means
are provided to urge said valve members, said valve cam
means forming an endless surface and rotating with said
rotary output member.

11.   The rotary engine according to Claim 10, and
in which said valve member for said intake manifold is
actuated by a first valve cam means and said valve member
for said exhaust manifolds is actuated by a second
valve cam means, said first and second valve cam means
being on opposite sides of said continuous cam.

12.   The engine according to Claim 3 and wherein
said pistons are positively biased outwardly so as to
ensure continuous engagement of the follower bearings with
the cam surface particularly during the intake stroke.

13. The engine according to Claim 3 and in which
said engine includes multiple generally parallel, in-line,
spaced apart cylinders with opposed pistons and each
cylinder having its own driven and valve cams.

14. The engine according to Claim 13 and in
which the driven cams for the individual cylinders are
angularly positioned with respect to each other so that
the mass of the rotary output member is balanced.

15. The engine according to Claim 3 and wherein
the expansion section from the last constant radius of
the top dead center area to the first or only radius at
bottom dead center is a harmonic curve setting forth the
radii as

$$R_\theta = S + [1-Cos\ (u)]$$

$$\text{where } u = \frac{\pi}{L}\ (\theta - k)$$

k = beginning radius point of the
harmonic curve (last constant radius
at top dead center)

m = ending radius point for the harmonic
curve (first or only radius at
bottom dead center)

L = 180 - (k + m)

S = 1/2 distance of travel of one piston

θ = angle of R from 0°

16. The engine according to Claim 15 and in which
the deceleration area of the expansion section in limit-
ation (a) is defined as a curve which continues to move the

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piston outwardly at a decelerating rate and which rate

decreases in reverse order of changes in the radii by

which the acceleration area increased.

17. The engine according to Claim 16 and in which

the expansion section may be of a greater or lesser amount

than 90° of rotation in order to extend the pistons to

bottom dead center position.

18. The engine according to Claim 17 and in which

the exhaust section is shaped after bottom dead center

position to return the pistons to top dead center position

at first slowly and then increases the speed so as to

minimize thrust loads on the piston rods and bearings.

19. The engine according to Claim 18 and in which

said driven cam surface is provided with predetermined

arcuate areas of constant radius from the axis of the rotary

member for radially opposed bottom dead center positions for

the pistons in order to allow for predetermined periods of

time and degrees of rotation at which the chamber between

the pistons is at constant volume at bottom dead center.

20. The rotary engine according to Claim 15 and

in which said intake and exhaust manifolds are opened to

said cylinder by ports and which ports are opened and closed

according to the position of the pistons.

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21. The rotary engine according to Claim 15 and
in which said intake and exhaust manifolds are provided
with openings which are provided with and opened and
closed by valve members actuated in accordance with the
requirements of the engine.

22. The rotary engine according to Claim 21 and
in which said valve members are provided with stem portions
which engage valve cam means against which resilient means
are provided to urge said valve members, said valve cam
means forming an endless surface and rotating with said
rotary output member.

23. The rotary engine according to Claim 22 and in
which said valve member for said intake manifold is
actuated by a first valve cam means and said valve member
for said exhaust manifolds is acutated by a second valve
cam means, said first and second valve cam means being on
opposite sides of said continuous cam.

24. The engine according to Claim 15 and wherein said
pistons are positively biased outwardly so as to ensure
continuous engagement of the follower bearings with the
cam surface particularly during the intake stroke.

25. The engine according to Claim 15 and in which
said engine includes multiple generally parallel, in -line,
spaced apart cylinders with opposed pistons and each
cylinder having its own driven and valve cams.

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26. The engine according to Claim 13 and in which the driven cams for the individual cylinders are angularly positioned with respect to each other so that the mass of the rotary output member is balanced.

27. A reciprocating, rotary, combustion engine, comprising:

a) a hollow, non-rotatable support block member of predetermined length and adapted to be supported in an operative position,

b) at least one cylinder member supported in said block member such that the ends of said cylinder open to the outside of said block member and whereby within said block member there is spaced externally of the cylinder, said cylinder including opposed pistons with piston rods extending out of the opposed ends of said cylinder and said piston rods being provided at their outer ends with follower bearing means for engaging a driven cam surface,

c) a rotary output member mounted for rotation around said support block member and including a generally round exterior surface and a continuous interior driven cam surface for said cylinder,

d) said cam being shaped such that it includes predetermined arcuate areas of constant radius from the axis of the rotary member at least for radially equal and opposed top dead center positions for said pistons in order to allow for predetermined periods of time and degrees of rotation at which the chamber between the pistons is at constant volume, said cam also providing at least radially equal and opposed bottom dead center positions, and said

B

cam further being shaped such that every radius therein
has an equal and opposite counterpart radius,

e) said cam being shaped after the top dead
center constant volume areas to provide an expansion section
having an acceleration area for moving the pistons away
from top dead center at an accelerating velocity and then
a deceleration area for moving the pistons away from top
dead center at a decelerating velocity toward bottom dead
center, and wherein the maximum value of the radius in the
acceleration areas cannot be greater than any point on a
line which is tangent to the last constant radius value
of the top dead center area, and

f) said cylinder being provided with means
for admitting fuel thereto, means for timed ignition of
the same and further provided with air intake and exhaust
gas manifolds and means for selectively opening and closing
said cylinder to said intake and exhaust manifolds.

28. The engine according to Claim 27 and in which
the cam is shaped after bottom dead center to provide
opposed return areas of a predetermined number of degrees
of rotation of the rotary output member for returning the
pistons to the next top dead center area.

29. The engine according to Claim 28 and in
which the deceleration area of the expansion section in
limitation (e) is defined as a curve which continues to
move the piston outwardly at a decelerating rate and which
rate decreases in reverse order of changes in the radii
by which the acceleration area increased.

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30. The engine according to Claim 29 and in which the expansion section may be of a greater or lesser amount than 90° of rotation in order to extend the pistons to bottom dead center position.

31. The engine according to Claim 30 and in which the areas of acceleration and deceleration may be of unequal degrees of rotation.

32. The engine according to Claim 28 and in which the expansion section from the last constant radius of the top dead center area to the first or only radius at bottom dead center is a harmonic curve setting forth the radii as

$$R_\theta = S + [1 - \cos(u)]$$

where $u = \dfrac{\pi}{L}(\theta - k)$

k = beginning radius point of the harmonic curve (last constant radius at top dead center)

m = ending radius point for the harmonic curve (first or only radius at bottom dead center)

L = 180 − (k + m)

S = 1/2 distance of travel of one piston,

θ = angle of R from 0°

33. The engine according to Claim 28 and in which the exhaust section is shaped after bottom dead center position to return the pistons to top dead center position at first slowly and then increases the speed so as to minimize thrust loads on the piston rods and bearings.

34. The engine according to Claim 33 and in which said driven cam surface is provided with predetermined arcuate areas of constant radius from the axis of the rotary member for radially opposed bottom dead center positions for the pistons in order to allow for predetermined periods of time and degrees of rotation at which the chamber between the pistons is at constant volume at bottom dead center.

35. The rotary engine according to Claim 28 and in which said intake and exhaust manifolds are opened to said cylinder by ports and which ports are opened and closed according to the position of the pistons.

36. The rotary engine according to Claim 28 and in which said intake and exhaust manifolds are provided with openings which are provided with and opened and closed by valve members actuated in accordance with the requirements of the engine.

37. The rotary engine according to Claim 36 and in which said valve members are provided with stem portions which engage valve cam means against which resilient means are provided to urge said valve members, said valve cam means forming an endless surface and rotating with said rotary output member.

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38.   The rotary engine according to Claim 37 and
in which said valve member for said intake manifold is
actuated by a first valve cam means and said valve member
for said exhaust manifold is actuated by a second valve
cam means, said first and second valve cam means being on
opposite sides of said continuous cam.

39.   The engine according to Claim 28 and wherein
said pistons are positively biased outwardly so as to ensure
continuous engagement of the follower bearings with the
cam surface particularly during the intake stroke.

40.   The engine according to Claim 28 and in which
said engine includes multiple generally parallel, in-line,
spaced apart cylinders with opposed pistons and each
cylinder having its own driven and valve cams.

41.   The engine according to Claim 40 and in
which the driven cams for the individual cylinders are
angularly positioned with respect to each other so that
the mass of the rotary output member is balanced.

FIG. 1

FIG. 2

FIG. 3

FIG. 4   FIG. 5

FIG.6   FIG. 7   FIG. 8   FIG. 9

FIG. 10

FIG. 11

FIG. 12

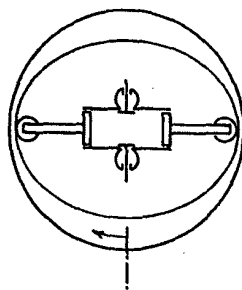# FIG. 13

FIG. 14D

POWER STROKE
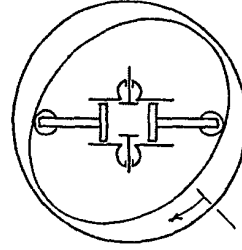
FIG. 14C
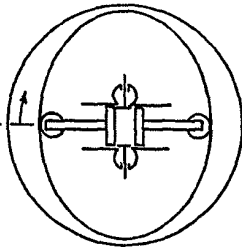
TOP DEAD CENTER
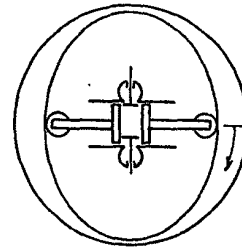MIXTURE IGNITED

FIG. 14B
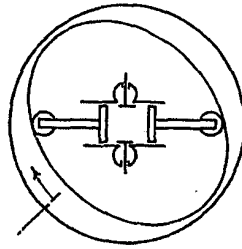
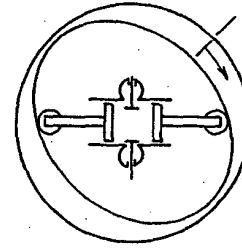COMPRESSION STROKE
BOTH VALVES CLOSED

FIG. 14A
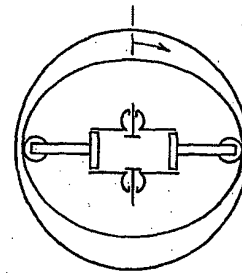
BOTTOM DEAD CENTER
START COMPRESSION
INTAKE VALVE CLOSING

FIG. 14H

INTAKE STROKE
INTAKE VALVE OPEN

FIG. 14G

TOP DEAD CENTER
BEGIN INTAKE

FIG. 14F

EXHAUST STROKE
EXHAUST VALVE OPEN

FIG. 14E

BOTTOM DEAD CENTER
START EXHAUST
EXHAUST VALVE OPEN

FIG. 15

FIG. 16

PRESSURE

VOLUME

TDC   BDC

P = 1.0 ATM

IDEAL

ACTUAL

SCHEMATIC IDEAL
SPARK IGNITION OTTO CYCLE

144, 146

148, 150

148, 150

Patent Agents
Smart & Biggar